Exhaust Temperature Control of Heavy Duty Gas Turbine Due to Incremental Load Demand

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Abstract - Industrial heavy duty gas turbines (HDGT) are specially designed gas turbines for power generation employed in critical industries such as power generation, oil and gas, process plants, aviation, as well as domestic and smaller related industries. HDGT automation, supervisory and control system are necessary for proper monitory and regulation of industrial production efficiency management. One of the major limitations to the performance of gas turbines is the inadequate control of the exhaust temperature. A peculiar problem in Trans-Amadi power plant, is the fact that increasing load tend to cause the exhaust temperature to rise to a level that leads to the shutdown of the gas turbine in operation. As a result of the frequent shutdown of the heavy duty gas turbine due to unusual temperature rise in proportion to the load increase, the Trans-Amadi power plant power plant like others experiences high fluctuation. The objective of the research is to improve the performance of the supervision and monitoring control of HDGT exhaust temperature. A controller is designed for a typical HDGT exhaust temperature control. The simulation results of the heavy duty gas turbine model and the controller showed that at 13seconds, desired settling time, the load torque drops to zero because gas turbine is now running on full load and it is steady. Also, when the power rises, the designed controller brings the load (or speed) to a steady state, that is 98% or more rated value at full load while regulating the exhaust temperature below rated value at a time of about 13seconds. The work provides an insight into control performance of a 25MW single shaft heavy duty gas turbine for temperature control like the one in Trans- Amadi power plant.

Key Words: Gas-Turbines, Exhaust, Temperature, Monitoring, Control

I INTRDUCTION

Gas turbines (GTs) are the significant and indispensable components of typical modern day industries especially in electricity generation. Gas Turbine engine is currently the most sought machine for the purpose of power generation and propulsion application [1]. The gas turbine is the major versatile item of turbo machinery used in several different modes in critical industries such as power generation, oil and gas, process plants, aviation, as well as domestic and smaller related industries [2]. A gas turbine is an internal combustion engine which uses the gaseous energy of air to convert chemical energy of fuel to mechanical energy [3]. It is mainly designed to extract energy from the fuel as much as possible. There are different components such as compressors, turbines, combustion chambers, etc., which constitutes a complex assembly of the gas turbine engine designed based on thermodynamic laws.

The compressor is a main section of a gas turbine cycle and compresses air for the combustion [4]. The compressor consists of the rotor and the stator mounted between bearings in a casing. Pressure is increased by each stage of the compressor multi-stage unit. Each stage is made up of one vertical layer of rotating blades, and stator vanes while the stator vanes decrease the air velocity and turn it into static pressure [4]. Ambient air is drawn through an inlet duct and passes through a filter before entering the compressor at the start of the compressor operation.

 Gas turbines are usually made up of an axial compressor, a combustion chamber and a turbine**.** The basic four steps for any internal combustion engine applicable to gas turbines are: intake of air, compression of the air, combustion (where fuel is injected and burned to convert the stored chemical energy to mechanical energy), expansion and exhaust (where the converted energy is put to use). Gas turbines have basically the same operation. In each operation cycle, air is drawn by the compressor into the engine through the inlet duct, which compresses it and then delivers it to the combustion chamber. Within the combustion chamber the air is mixed with fuel and the mixture is ignited, producing a rise in temperature and hence an expansion of the gases [5]. These gases are exhausted through the engine nozzle by first passing through the turbine, which extracts enough mechanical energy from them to keep the compressor rotating and to keep the engine in self-sustaining.

Gas turbine engines can be classified into several types based on its applications and work. Broad classification can be made based on the number of engine shafts, either single shaft or twin shaft engine [1]. A single shaft gas turbine engine has the compressor, turbine and load on the same shaft while in twin shaft engines, the compressor and the gas generator turbine are on the same shaft whereas the power turbine and loads are on a different shaft. Other types of gas turbine engine based on the work are the Jet engines, Industrial gas turbines for mechanical drive, micro-turbines, aero derivative gas turbine, radial gas turbines, heavy duty gas turbines etc.

Industrial heavy duty gas turbines (HDGT) are specially designed gas turbines for power generation which are specified by their long life and higher availability compared to other types of gas turbines [5]. The major parts of a HDGT include the inlet duct, the combustion chamber, the compressor, the turbine and the nozzle (gas-deflector). High

thermal efficiency and high power output of heavy duty gas turbine engines have made them very effective and useful in power generation [6]. HDGTs are important and widely prime movers in the power sector of most countries.

A typical heavy duty gas turbine has many different components distributed at far distance from one another. Each of these components has a partially independent control system. In order to monitor and supervise the activities of these components, a control system that integrates the separate component controls is required. Finding optimal and lasting improvements to HDGT control systems for efficient, reliable and durable functionality has been one of the major concerns of researchers in recent times. One the most suitable control system for modern heavy duty gas turbines is the distributed control system.

A Distributed Control System (DCS) is a flexible hardware and software control system that is easy to modify and configure to handle a large number of controller loops with high performance and optimization. Typical DCS systems are made up of local control unit local controllers, data acquisition unit, central operator display, data/signal highway etc. DCS systems allow different modes of process control such as manual, auto, supervisory for each local control loop and implementation of digital control algorithms. The control loop of a DCS is same as the conventional feedback control loop, but with additional digital components. The digital components (computers) are used to carry out all of the control calculations. One major application of DCS is in the Supervisory and Monitoring Unit of an industrial Control System for heavy duty plants.

Supervisory and Monitoring Unit is an industrial Control System (ICS) that can supervise and monitor industrial processes in a power plant. Industrial processes include those of manufacturing, production, power generation, fabrication, refining etc and may run in continuous, batch, repetitive or discrete modes [7]. Industrial Control systems are computer-based systems that monitor and control industrial processes that exist in the physical world.

Supervisory Control in a plant comes into play when the control system put in place to control field devices and facilities like sensors, actuators, Motor Control Center (MCC) need supervision and monitoring.

The supervisory control and monitoring unit is pure plant control which supervises and monitor the entire plant [7]. Supervisory and monitoring unit has a supervisory controller (control server) that communicates to its subordinates via a control network.

Supervisory and monitoring unit is a control system used in plant to supervise, communicate, control, monitor field devices, plant, facilities and equipment. Unit control for a gas turbine monitors, protects and control the gas turbine machine/ auxiliaries . Supervisory and monitoring unit will supervise the various control systems and monitor every device, facility/machine that makeup the plant.

In this research, the controller is designed for a typical heavy duty gas turbine exhaust temperature control for a 25MW turbine. The aim of the research is to improve the performance and broaden the scope of Exhaust Temperature Control of heavy duty gas turbines.
Whenever the need arises to control and monitor more than one variable or parameter the supervisory control comes into play.
Several research have been carried out in this area to improve the existing control system of heavy duty gas turbine. This is to solve the peculiar problem of rise in exhaust temperature that is not steady and proportional to the rise in load that might arise from time to time. A typical case of this problem can be found in the most power plant operational in some developing areas where increasing load tend to cause the exhaust temperature to rise to a level that leads to the shutdown of the running gas turbine. As a result of the frequent shutdown of the heavy duty gas turbine due to unusual temperature rise, the many of these power generation plants experiences high fluctuation. Moreover, if this problem is allowed to linger longer, the life span of the plant could be shortened. Therefore, there is urgent need for the exhaust temperature to be properly controlled to ensure that it is maintained at an appropriate level. This requires an improved supervisory and monitoring control of the major components of the gas turbine especially the exhaust temperature. The main objective of the study is to design an exhaust temperature controller for heavy duty gas turbines.

II DYNAMIC EQUATIONS OF THE GAS TURBINE

Fig. 1 shows a simplified diagram of a gas turbine model . A typical gas turbine model has basically three control loop as shown but only two out of the three loops will be considered in this context.
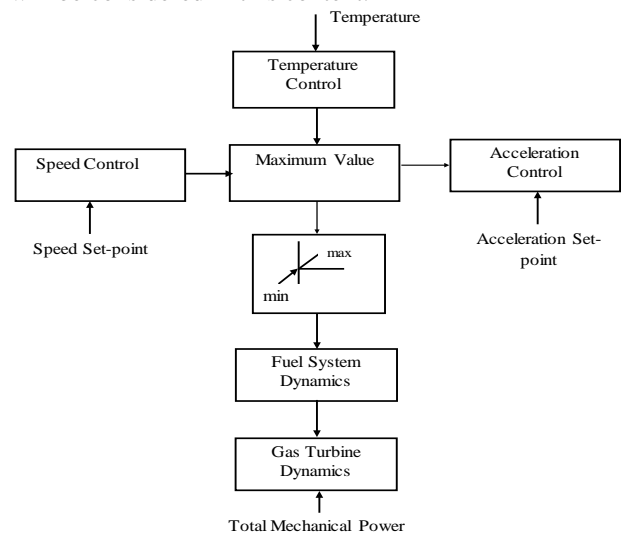

Fig. 1 A typical model of a gas turbine

AThe Fuel System
This unit comprises the fuel valve and actuator. Fuel injection into a gas turbine is determined by the valve positioner whose activity is controlled by the speed controller. A typical valve positioner transfer function is given as:

$$V(s) = \frac{a}{bs+c} \qquad (1)$$

where a, b, and c are the valve positioner constants.
The fuel system actuator transfer function is:

$$F(s) = \frac{1}{T_{fc}s+1} \qquad (2)$$

where T_{fc}, is the fuel system actuator time constant in seconds.

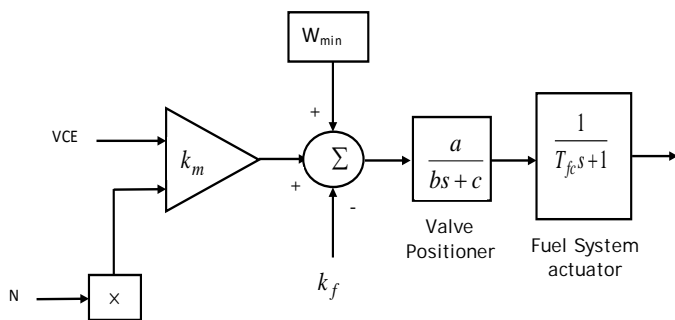The Simulink block diagram of the fuel system is represented in fig. 2 below.

Fig. 2: Simulink Block of the fuel system

VCE is the output of the least value gate (LVG) that governs the least amount of fuel needed for a given operating point and also an input to the fuel system. N is the per unit turbine speed which is also an input to the fuel system. W_{min} is the minimum amount of fuel flow. k_m is equal to 1-W_{min} and k_f is the fuel system feedback.

B   Compressor-turbine Dynamics
The compressor-turbine is often referred to as the heart of the gas turbine. It has a small transport delay associated with the combustion reaction time given in Equation (3), a time lag given as expression which associated with the compressor discharge volume and transport delay given as expression, and for transport of gas from the combustion system through the turbine.

The burning of the fuel in the combustor is presented by the following function:

$$C_{TD} = e^{-sT_{CR}} \qquad (3)$$

where T_{CR} is the combustion reaction time delay constant in seconds.
The transfer function of the hot computation gas expansion is expressed as follows:

$$T(s) = \frac{1}{T_{CD}S+1} \qquad (4)$$

where T_{CD} is the compressor discharge volume time constant in seconds.
The compressor-turbine transport delay is given as:

$$CT_{td} = e^{-sT_{CR}} \qquad (5)$$

The mechanical torque in Nm produced, which drives the electric generator is presented by the following equation .

$$T_m = A + B\dot{m}_f + C(1-N) \qquad (6)$$

The exhaust temperature in (°C) is given as :

$$T_{EX} = T_R - D(1-\dot{m}_f) + E(1-N) \qquad (7)$$

A and B are the coefficients of output torque which could be obtained by applying the actual data in Table (1). N is the per unit rotor speed. The value for C in the torque Equation (7) varies between 0.5 and 0.67 for heavy duty gas turbine (HDGT) . In this context, C is assigned a value of 0.5. D and E are the coefficients of the exhaust temperature.

The input to this subsystem is the per unit (p.u.) fuel demand signal W_f and outputs are the p.u. turbine torque and exhaust temperature (°C).

C   *Case Study of Gas the Turbine*
The Trans Amadi gas turbine considered in this study is heavy duty single shaft gas turbine with installed capacity of 26MW and very high rotational speeds between 1200 rpm and 1500 rpm. The output voltage and currents are 11KV and 1267A. Power factor and ambient temperature rating are 0.8 and 25-45 degree Celsius. Table 1 shows the nominal data or design specifications and a typical average operational data presented in Table 2 of the selected heavy duty gas turbine (HDGT).

Table 1: Nominal data of Selected HDGT

Parameters	Symbol	Unit	Value
Electrical power output	P_{GT}	MW	26.3MW
Heat Rate	H_R	kJ/kwh	12650
Exhaust Temperature	T_{OE}	°C	487
Exhaust Mass Flow	$\dot{m}_{no}$	Kg/s	124.1
Pressure Ratio	PR	-	9.87
Normal speed	-	rpm	5100
Lower Heating Value of Fuel(LHV)	H	kJ/kg	43309

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Table 2: Average operating data for Trans-Amadi gas turbine

Component	Parameter	Unit	Value
Compressor	Inlet temperature	℃	30.4
	Outlet temperature	℃	367
Combustion chamber	Inlet pressure	bar	1.013
	Outlet pressure	bar	10
	Mass flow rate (Air)	Kg/s	122.9
	Fuel consumption (flow rate)	Kg/s	1.2
Gas turbine	Inlet temperature	℃	959
	Outlet temperature	℃	487
Other data	Exhaust gas flow (flow rate of gas)	Kg/s	124.1
	GT power output	MW	25

III TURBINE AND GAS TURBINE PARAMETERS

In order to compute the values of the turbine and compressor efficiencies, the values of the nominal data and operational data of Tables 1 and 2 are used based on the detailed approach and equations in as follows:

A Turbine efficiency (n_t):

$$x_{h(oc)} = \left(PR \times \frac{\dot{m}_{oc}}{\dot{m}_{no}} \right)^{\frac{\gamma_h - 1}{\gamma_h}} = \left(9.87 \times \frac{124.1}{124.1} \right)^{\frac{1.33 - 1}{1.33}} \tag{8}$$

$$T_{4s(oc)} = \frac{T_{3(oc)}}{x_{h(oc)}} = \frac{959 + 273}{1.76} = 700K = 427 \tag{9}$$

℃

Where $x_{h(oc)}$, $\dot{m}_{oc}$, $\dot{m}_{no}$, γ_h are the ratio of input-output temperatures for isentropic process, exhaust mass flow rate, and specific heat ratio at hot end (combustor, turbine). The index (*oc*) and (*no*) mean operating conditions and nominal conditions. $T_{4s(oc)}$ and $T_{3(oc)}$ are exhaust temperature and turbine inlet temperature at operating conditions.

$$\eta_t = \frac{T_{3(oc)} - T_4}{T_{3(oc)} - T_{4s(oc)}} = \frac{959 - 487}{959 - 427} = \frac{472}{532} = 0.89 \tag{10}$$

B Compressor Efficiency (n_c):

$$x_{c(oc)} = \left(PR \times \frac{\dot{m}_{oc}}{\dot{m}_{no}} \right)^{\frac{\gamma_c - 1}{\gamma_c}} = \left(9.87 \times \frac{124.1}{124.1} \right)^{\frac{1.4 - 1}{1.4}} = 1.92 \tag{11}$$

$$T_{2s(oc)} = T_{1(oC)} \times x_{c(oc)} \tag{3.12}$$

$$T_{2s(oc)} = (26 + 273) \times 1.92 = 574.08K = 301.08$$
℃

The compressor outlet air temperature which is given (Table 2) by:

$$T_{2(oc)} = 367 \text{ ℃}$$

$$\eta_c = \frac{T_{2s(oc)} - T_1}{T_{2(oc)} - T_1} = \frac{301.08 - 26}{367 - 26} = .0.81 \tag{13}$$

Where $x_{c(oc)}$, γ_c are the ratio of input-output temperatures for isentropic compression, exhaust mass flow rate, and specific heat ratio at hot end (combustor, turbine). $T_{2s(oc)}$ and $T_{2(oc)}$ are compressor discharge temperature and compressor outlet temperature at operating conditions. T_1 is the ambient temperature. η_{comb} is the combustion efficiency. A value of 0.99 is assumed for combustion system. Since it is near unity, it has been chosen in this context as unity. C_{ph} is the specific heat of hot end (turbine)

Parameters will be computed for mechanical power block in equation (6) based on nominal value:

$$x_h = (PR)^{\frac{\gamma_h - 1}{\gamma_h}} = (9.87)^{\frac{1.33 - 1}{1.33}} = 1.76 \tag{14}$$

$$x_c = (PR)^{\frac{\gamma_c - 1}{\gamma_c}} = (9.7)^{\frac{1.4 - 1}{1.4}} = 1.93 \tag{15}$$

The value of A and B in Equation (6) can be extracted as follows

$$A = \frac{\dot{m}_n T_1}{P_{Gn}} \left\{ C_{ph} \times \eta_t \left(1 - \frac{1}{x_h} \right) - \frac{x_c - 1}{x_c} \times \left[C_{pc} - C_{ph} \times \eta_t \left(1 - \frac{1}{x_h} \right) \right] \right\} \tag{16}$$

$$A = \frac{124.1 \times (273 + 15)}{26300} \times \left\{ \begin{array}{l} 1.1569 \times 0.89 \left(1 - \frac{1}{1.76} \right) - \frac{1.93 - 1}{0.81} \\ \times \left[1.0047 - 1.1569 \times 0.89 \left(1 - \frac{1}{1.76} \right) \right] \end{array} \right\} = -0.2685$$

$$B = \frac{\eta_{comb} \times \eta_t \times \dot{m}_{fn}}{P_{Gn}} \left[\left(1 - \frac{1}{x_h} \right) \right] \tag{17}$$

$$B = \frac{1 \times 0.89 \times 43309 \times \dot{m}_{fn}}{26300}\left(1 - \frac{1}{1.76}\right) = 0.6329\dot{m}_{fn}$$

$$\dot{m}_{fnpu} = \frac{P_{Gpu} - A}{B} = \frac{1 + 0.2685}{0.6329\dot{m}_{fn}} = 1 \Rightarrow \dot{m}_{fn} = 2.004 kg / s$$

(18)

$$\Rightarrow B = 1.2683$$

where $\dot{m}_n$ and $\dot{m}_{fn}$ are the air and fuel nominal flow rates, P_{Gn} and P_{Gpu} are the nominal output power and the per unit (p.u.) output power which is equal to the p.u. torque.

It should be noted that the values of C_{pc}, C_{ph}, γ_c, and γ_h are assumed as 1.0047kJ/kgK, 1.1569kJ/kgK, 1.4, and 1.33, as a common approach to be employed for the cold end and hot end air properties

In order to determine the turbine exhaust temperature, the exhaust temperature parameters D and E are computed using nominal flow.

$$D = \eta_{comb}\frac{H}{C_{ph}} \times \frac{\dot{m}_{fn}}{\dot{m}_n}\left[1 - \left(1 - \frac{1}{x_h}\right)\eta_t\right]$$ (19)

$$D = 1 \times \frac{43309}{1.1569} \times \frac{2.004}{124.1}\left[1 - \left(1 - \frac{1}{1.76}\right) \times 0.89\right] = 372.19\ ^oC$$

The exhaust temperature coefficients E varies in the range of 0.55 to 0.65 of the rated temperature. In this context, a value of: $E = 0.6T_R = 0.6 \times 487 = 292.2\ ^oC$. Equations (8) to (3.19) present the analysis of the considered gas turbine parameters.

IV   IMPLEMENTED CONTROLLER DESIGN

In this research, two controllers are designed and implemented using Matlab software. They are proportional integral and derivative (PID) controller and analogue compensator. The PID controller is designed and integrated into the gas turbine control loop to regulate the gas turbine speed. The analogue compensator is integrated into the control loop to regulate the gas turbine exhaust temperature.

The controller design used in this context is to use the Matlab/Simulink optimization toolbox. The selected gains of the PID controller are constrained to ensure that the performance specifications of the system are attained as presented in Table 3.

Table 3: Performance specification

Performance condition	Value
Rise time (s)	0.2
Settling time (s)	10
% Overshoot	5
% Rise	90

A Speed Control

In order to design and implement the PID controller for speed control, a nonlinear Simulink PID block is used. The gains selection was achieved by constraining the response of the system to meet design requirement as in Table 3.

Table 4: The gains of the designed PID controller

Parameter	Gain
k_p	936.330
k_i	0.062190
k_d	102.985
N	10

$$U(s) = K + \frac{I}{s} + \frac{DNs}{s + N}$$ (20)

where $K = k_p E(s) = 10$, $I = k_i E(s) = 0.001$, $D = k_d E(s) = 1.656$, and $N = 10$, which is the filter coefficient. This yields:

$$U(s) = 10 + \frac{0.001}{s} + \frac{1.656s}{s + 1}$$ (21)

B Temperature Control

In order to design the analogue compensator for the temperature control, the approach used in designing the compensator for the speed control is employed as well. The designed temperature control compensator is given in Equation (22).

$$Gc(s) = \frac{T_c s + 1}{T_t s}$$ (22)

where $T_c = 3.3$ and $T_t = 150^O$ C

V   RESULTS OF SIMULATION

The results obtained from the simulations performed for the supervisory control in Matlab/Simulink environment for a single shaft heavy duty gas turbine are shown below. Fig. 3 through 7 show the performance of the gas turbine under the supervisory control at nominal operating condition. Fig. 3 shows the speed/load frequency transient performance of the turbine when it is operating at step speed (1p.u.) or 100% full load.   Fig. 4 shows the exhaust performance characteristic at nominal rated load. Fig. 5 show the robust performance of the supervisory control action. Fig. 6 represents the fuel demand plot at nominal operating condition. Fig. 7 shows the load torque characteristics.

Two basic control loops introduced for the control action were considered in this context. The first one is the speed control loop, which integrates a proportional integral and derivative (PID) speed controller. The second one is the temperature control loop which integrates analogue temperature compensator.

Fig. 3 shows the speed control loop performance at nominal load (100% full load). This control determines the fuel demand according to the load reference and the rotor speed deviation. It can be seen that the PID speed controller is dominant and operates on the speed error formed between the reference speed (1 p.u.) and the rotor speed of the gas turbine. It reaches the rated speed of (1 p.u. or 100%) at nominal load with a settling time of 10seconds with an overshoot of 5%.

It can be seen from Table 4.1 that during start-up operation, the exhaust temperature rises to a temperature of about 900 °C compare to 700 °C from the site and then the temperature controller is activated into control action. The controller regulates the exhaust temperature and brings it to an improved temperature of about 200 °C at about 10 seconds and then settles. The temperature at which the exhaust is regulated is compared to the site temperature of 488 - 510 °C. This shows that designed temperature controller improved the gas turbine exhaust temperature performance.

Fig.4 presents the exhaust temperature control for the supervisory system. This control prevents the exhaust temperature not to damage the gas turbine. When the temperature of the exhaust tends to rise closer to the referenced value ,the temperature control compensator acts on the breed/air valves and the extraction motors/fans to increase the airflow so as to decrease the exhaust gas temperature. This control reacts by increasing or decreasing flow exhaust temperature in a smooth, steady and proportional to the load demand of the turbine. It should be noted that reference temperature is defined by the supervisory control for the exhaust temperature. This control reacts by decreasing the reference temperature when gas turbine inlet temperature exceeds its nominal value. Hence, as the load is increasing to its rated nominal value (1 p.u.), the exhaust temperature rises and this can cause overheating of the gas turbine thereby making it to shut down. In order to take care of this, the supervisory control action ensures that at increasing load, the exhaust temperature is quickly brought to a level lower than the rate value of the gas turbine at about 10seconds.

In Fig. 5, the action of the temperature control and speed control at the low value selector (LVS) is presented. The lower value is selected by the LVS, and it determines the fuel flow. These inputs determine the fuel demand signal. As it can be seen, during increasing load, a quick increase of the fuel demand signal is observed (see fig.6). Hence, soon after the load (or speed) has stopped increasing and maintain a stable state, LVS function activates the temperature control to avoid exhaust temperature overheating. This allows increase of power production with constant temperature. In this way the system return to stable operation of speed (or load-frequency) control and maintains a steady state at 10seconds.

In Fig.7, the load torque increases as the speed (or load) increases. At about 10seconds, which is desired settling time, the load torque drops to zero because gas turbine is now running on full load and it is steady. Generally it can be seen that the exhaust temperature controller designed is able to achieve the main objective of this work which is to regulate or control the problem of shut down of about 19MW which is not equivalent to the rated power at full load. When the load rises, the exhaust controller brings the temperature to a steady stated, that is 100% or 1 p.u. rated value at full load while reducing the exhaust temperature below rated value at about 10seconds.
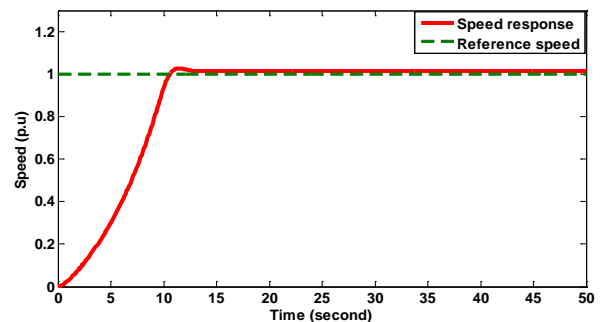

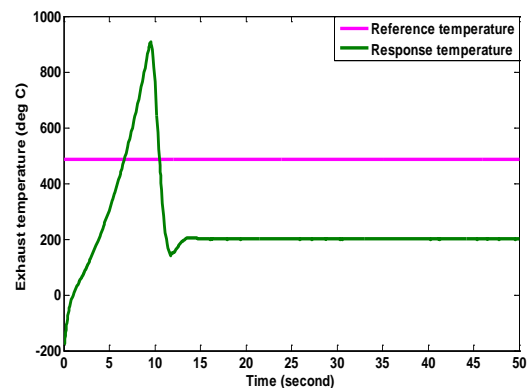Fig. 3: Speed control transient response performance

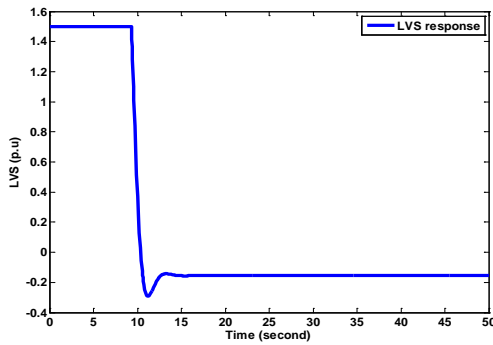Fig. 4: Exhaust temperature performance at nominal load

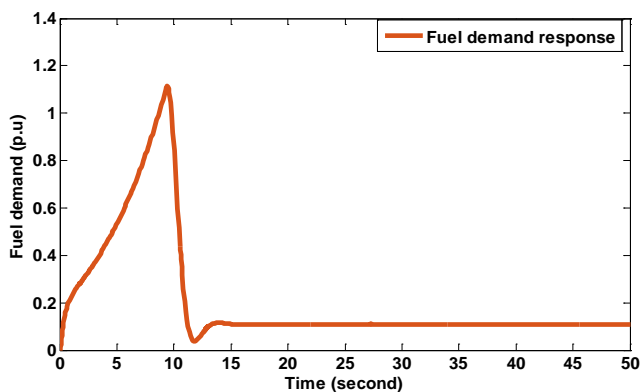Fig. 5 Low value selector (LVS) performance response
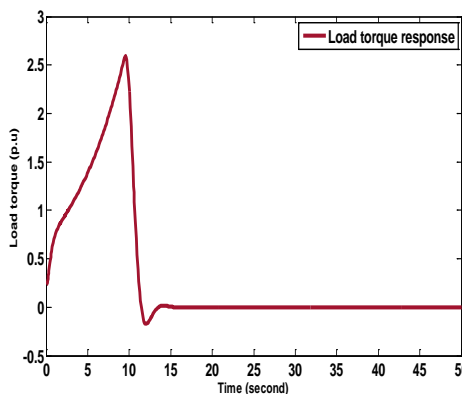

Fig 6: Fuel demand performance response

Fig. 7: Load torque performance response at nominal load

VI   CONCLUSION

This research has presented design of controller unit for the exhaust control a gas turbine plant. The work was aimed at designing a control unit for regulating the speed and exhaust temperature of the gas turbine plant. In order to achieve the objective of this work, mathematical equations representing the dynamics of the gas turbine system were obtained and transformed into their equivalent Simulink blocks. Nonlinear proportional integral and derivative (PID) controller and analogue compensator were designed to form the control loops representing exhaust temperature control unit of the gas turbine. In order to design these controllers, nonlinear Simulink block of PID was used and the system constrained to meet the design specifications. The analogue

compensator was designed by selecting appropriate parameters to form the temperature control. The designed controllers were integrated with gas turbine model dynamics to form the exhaust temperature control loop. The simulation parameters were obtained from the rated nominal operational data of the gas turbine and the operational data obtained from the considered gas turbine. The entire gas turbine supervisory control loop was implemented in Simulink. The simulation results obtained show that at nominal speed (1 p.u.), the exhaust temperature over heating was taken care of at about 10 seconds settling time at which point the system stabilizes.

The system performance shows that the exhaust temperature controller implemented in this context was able to take care of the problem in the considered industrial heavy duty gas turbine and as such the control can be applied in equivalent gas turbine of similar gas turbine.

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